Renewable and Sustainable Energy Reviews 14 (2010) 1187–1201

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

The potential of solar-driven humidification–dehumidification desalination for small-scale decentralized water production

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ARTICLE INFO

Article history:
Received 25 May 2009
Received in revised form 4 November 2009
Accepted 7 November 2009

Keywords:
Humidification
Dehumidification
Desalination
Decentralized water production
Solar power
Small-scale water production

ABSTRACT

World-wide water scarcity, especially in the developing world, indicates a pressing need to develop inexpensive, decentralized small-scale desalination technologies which use renewable resources of energy. This paper provides a comprehensive review of the state-of-the-art in one of the most promising of these technologies, solar-driven humidification–dehumidification (HDH) desalination. Previous studies have investigated many different variations on the HDH cycle. In this paper, performance parameters which enable comparison of the various versions of the HDH cycle have been defined and evaluated. To better compare these cycles, each has been represented in psychometric coordinates. The principal components of the HDH system are also reviewed and compared, including the humidifier, solar heaters, and dehumidifiers. Particular attention is given to solar air heaters, for which design data is limited; and direct air heating is compared to direct water heating in the cycle assessments. Alternative processes based on the HDH concept are also reviewed and compared. Further, novel proposals for improvement of the HDH cycle are outlined. It is concluded that HDH technology has great promise for decentralized small-scale water production applications, although additional research and development is needed for improving system efficiency and reducing capital cost.

Contents

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doi:10.1016/j.rser.2009.11.014

1. Introduction

The Millennium development goals set by the United Nations highlight the critical need of impoverished and developing regions of the world to achieve self-sustenance in potable water supply [1]. Desalination systems are essential to the solution of this problem. However, conventional desalination technologies are usually large-scale, technology intensive systems most suitable for the energy rich and economically advanced regions of the world. They also cause environmental hazards because they are fossil-fuel driven and also because of the problem of brine disposal. In the following sections these conventional desalination technologies are introduced and their drawbacks are discussed.

1.1. Conventional desalination technologies

Desalination of seawater or brackish water is generally performed by either of two main processes: by evaporation or by use of a semi-permeable membrane to separate fresh water from a concentrate. The most important of these technologies are listed in Table 1. In the phase-change or thermal processes, the distillation of seawater is achieved by utilizing a heat source. The heat source may be obtained from a conventional fossil-fuel, nuclear energy or from a non-conventional source like solar energy or geothermal energy. In the membrane processes, electricity is used either for driving high-pressure pumps or for establishing electric fields to separate the ions.

The most important commercial desalination processes [2] based on thermal energy are multi-stage flash (MSF) distillation, multiple effect distillation (MED) and vapor compression (VC), in which compression may be accomplished thermally (TVC) or mechanically (MVC). The MSF and MED processes consist of many serial stages at successively decreasing temperature and pressure. The MSF process is based on the generation of vapor from seawater or brine due to a sudden pressure reduction (flashing) when seawater enters an evacuated chamber. The process is repeated stage-by-stage at successively decreasing pressures. Condensation of vapor is accomplished by regenerative heating of the feed water. This process requires an external steam supply, normally at a temperature around 100 °C. The maximum operating temperature is limited by scale formation, and thus the thermodynamic performance of the process is also limited. For the MED system, water vapor is generated by heating the seawater at a given pressure in each of a series of cascading chambers. The steam generated in one stage, or "effect," is used to heat the brine in the next stage, which is at a lower pressure. The thermal performance of these systems is proportional to the number of stages, with capital cost limiting the number of stages to be used. In TVC and MVC systems, after vapor is generated from the saline solution, it is thermally or mechanically compressed and then condensed to generate potable water.

Table 1
Desalination processes.

Phase-change processes	Membrane processes
1. Multi-stage flash (MSF)	1. Reverse osmosis (RO)
2. Multiple effect distillation (MED)	2. Electrodialysis (ED)
3. Vapor compression (VC)	
4. Solar stills	

The second important class of industrial desalination processes uses membrane technologies. These are principally reverse osmosis (RO) and electrodialysis (ED). The former requires power to drive a pump that increases the pressure of the feed water to the desired value. The required pressure depends on the salt concentration of the feed. The pumps are normally electrically driven [3]. The ED process also requires electricity to produce migration of ions through suitable ion-exchange membranes [4]. Both RO and ED are useful for brackish water desalination; however, RO is also competitive with MSF distillation processes for large-scale seawater desalination.

The MSF process represents more than 90% of the thermal desalination processes, while RO process represents more than 80% of membrane processes for water production. MSF plants typically have capacities ranging from 100,000 to almost 1,000,000 m^3/day [5]. The largest RO plant currently in operation is the Ashkelon plant, at 330,000 m^3/day [6].

Other approaches to desalination include processes like the ion-exchange process, liquid–liquid extraction, and the gas hydrate process. Most of these approaches are not generally used unless when there is a requirement to produce high purity (total dissolved solids $<$10 ppm) water for specialized applications.

Another interesting process which has garnered much attention recently is the forward osmosis process [7]. In this process, a carrier solution is used to create a higher osmotic pressure than that of seawater. As a result the water in seawater flows through the membrane to the carrier solution by osmosis. This water is then separated from the diluted carrier solution to produce pure water and a concentrated solution which is sent back to the osmosis cell. This technology is yet to be proven commercially.

1.2. Limitations of conventional technologies

Conventional processes like MSF and RO require large amounts of energy in the form of thermal energy (for MSF) or electric power (for RO). Most desalination plants using these technologies are fossil-fuel driven. This results in a large carbon footprint for the desalination plant, and sensitivity to the price and availability of oil. To avoid these issues, desalination technologies based on renewable energy are highly desirable.

Solar energy is the most abundantly available energy resource on earth. Solar desalination systems are classified into two main categories: direct and indirect systems. As their name implies, direct systems use solar energy to produce distillate directly using the solar collector, whereas in indirect systems, two sub-systems are employed (one for solar power generation and one for desalination). Various solar desalination plants in pilot and commercial stages of development were reviewed by [8].

In concept, solar energy based MSF and MED systems are similar to conventional thermal desalination systems. The main difference is that in the former, solar energy collection devices are used. Some proposals use centralized, concentrating solar power at a high receiver temperature to generate electricity and water in a typical large-scale coproduction scheme [9]. These solar energy collectors are not yet commercially realized. It should be noted that at lower operating temperatures, solar collectors have higher collection efficiency, owing to reduced losses, and also, can be designed to use less expensive materials.

Moreover, owing to their fossil-fuel dependence, conventional desalination techniques are less applicable for decentralized water production. Decentralized water production is important for regions which have neither the infrastructure nor the economic resources to run MSF or RO plants and which are sufficiently distant from large-scale production facilities that pipeline distribution is prohibitive. Many such regions are found in the developing world in regions of high incidence of solar radiation. The importance of decentralizing water supply was reviewed in detail by [10].

For small-scale applications (from 5 to 100 m^3/day water production), the cost of water production systems is much higher than for large-scale systems. For RO systems, which are currently the most economical desalination systems, the cost of water production can go up to US$ 3/m^3 [11] for plants of smaller capacity. Also, RO plants require expert labor for operation and maintenance purposes. This a clear disadvantage for small-scale applications in less developed areas, particularly when compared to the HDH system.

2. Humidification–dehumidification (HDH) desalination technology

Nature uses solar energy to desalinate ocean water by means of the rain cycle (Fig. 1). In the rain cycle, seawater gets heated (by solar irradiation) and humidifies the air which acts as a carrier gas. Then the humidified air rises and forms clouds. Eventually, the clouds 'dehumidify' as rain. The man-made version of this cycle is called the humidification–dehumidification desalination (HDH) cycle.

The HDH cycle has received much attention in recent years and many researchers have investigated the intricacies of this technology. It should be noted here that the predecessor of the HDH cycle is the simple solar still. Several researchers [12–14] have reviewed the numerous works on the solar still and hence, this paper will not discuss that technology. However, it is important to understand the disadvantages of the solar still concept.

The most prohibitive drawback of a solar still is its low efficiency (gained-output-ratio less than 0.5) which is primarily the result of the immediate loss of the latent heat of condensation through the glass cover of the still. Some designs recover and reuse the heat of condensation, increasing the efficiency of the still. These designs (called multi-effect stills) achieve some increase in the efficiency of the still but the overall performance is still relatively low. The main drawback of the solar still is that the various functional processes (solar absorption, evaporation, condensation, and heat recovery) all occur within a single component. By separating these functions into distinct components, thermal inefficiencies may be reduced and overall performance improves. This separation of functions is the essential characteristic of the HDH system. For example, the recovery of the latent heat of condensation, in the HDH process, is affected in a separate heat exchanger (the dehumidifier) wherein the seawater, for example, can be pre-heated. The module for solar collection can be optimized almost independently of the humidification or condensation component. The HDH process, thus, promises higher productivity due to the separation of the basic processes.

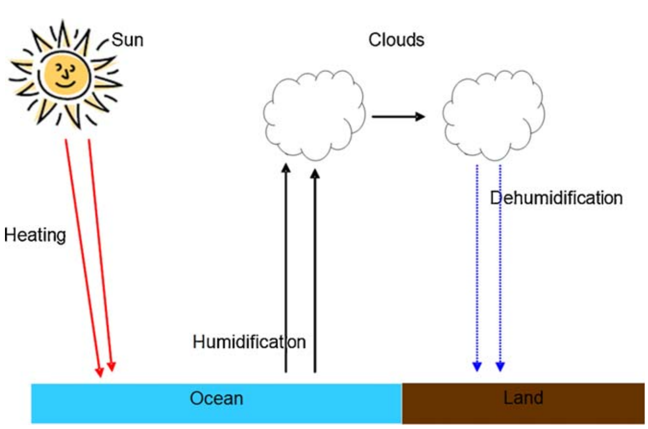


Fig. 1. Rain cycle.

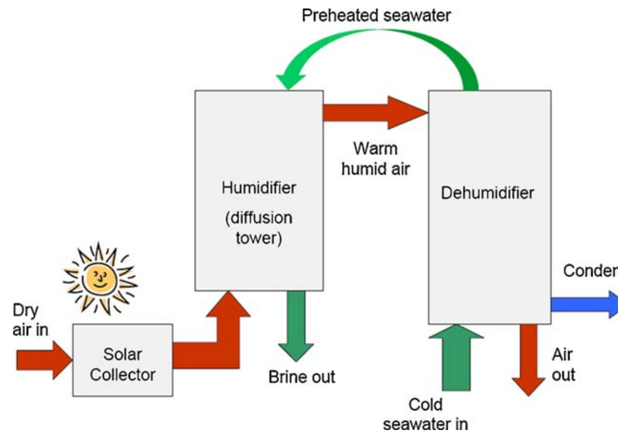


Fig. 2. A simple humidification–dehumidification (HDH) process.

The simplest form of the HDH process is illustrated in Fig. 2. The process consists of three sub-systems: (a) the air and/or the water heater, which can use various sources of heat like solar, thermal, geothermal or combinations of these; (b) the humidifier or the evaporator; and (c) the dehumidifier or the condenser.

2.1. Classification

HDH systems are classified under three broad categories. One is based on the form of energy used such as solar, thermal, geothermal, or hybrid systems. This classification brings out the most promising merit of the HDH concept: the promise of water production by use of low-grade energy, especially from renewable resources.

The second classification of HDH processes is based on the cycle configuration (Fig. 3). As the name suggests, a closed-water open-air (CWOA) cycle is one in which the air is heated, humidified and partially dehumidified and let out in an open cycle as opposed to a closed-air cycle wherein the air is circulated in a closed loop between the humidifier and the dehumidifier. The air in these systems can be circulated by either natural convection or mechanical blowers.

It is of pivotal importance to understand the relative technical advantages of each of these cycles and choose the one that is best in terms of efficiency and cost of water production. In the published literature, not much attention has been paid to optimization of the cycle itself as compared to the optimization of the three sub-systems. Further, a few investigators [15–17] have studied the cost of the HDH cycles and found that the cost of water production is high. This high cost may be brought down to more reasonable levels by understanding and optimizing the overall cycle. This optimization is a focus of the remainder of this paper.

The third classification of the HDH systems is based on the type of heating used: water- or air-heating systems. The performance of the system depends greatly on whether the air or water is heated. While there are many decades of experience and wisdom on solar water-heating devices, relatively little work has been done on the

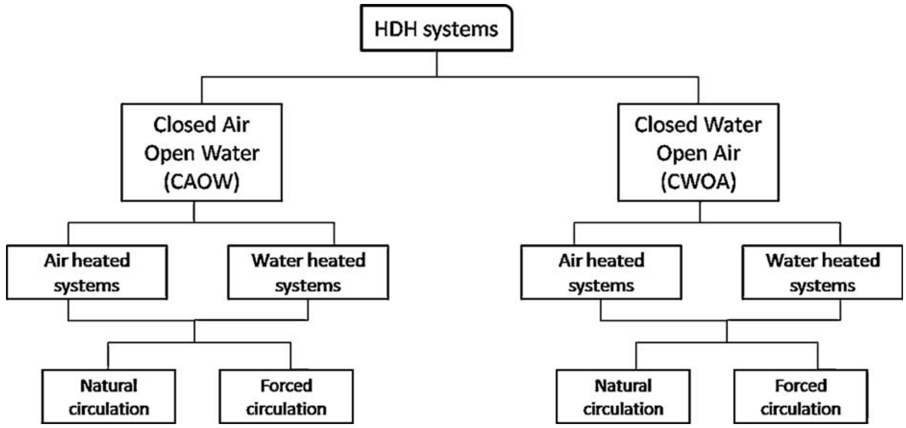


Fig. 3. Classification of typical HDH processes (based on cycle configuration).

solar collectors for air heating. Considering their importance to the overall HDH system performance, solar air-heating devices are also reviewed in this paper.

3. Review of systems in literature

As a first step for understanding different works in literature the following performance parameters are defined.

(1) *Gained-Output-Ratio* (*GOR*): the ratio of the latent heat of evaporation of the distillate produced to the total heat input absorbed by the solar collector(s). This parameter is, essentially, the efficiency of water production and an index of the amount of the heat recovery effected in the system. This parameter does not account for the solar collector efficiency as it just takes into account the heat obtained in the solar collector. For the HDH systems to have thermal performance comparable to MSF or MED, a GOR of at least 8 (corresponding to energy consumption rates of ~300 kJ/kg) should be achieved.
(2) *Specific water production*: the amount of water produced per m^2 of solar collector area per day. This parameter is an index of the solar energy efficiency of the HDH cycle. This parameter is of great importance as the majority of the capital cost of the HDH system is the solar collector cost: 40–45% for air-heated systems [18] and 20–35% for water-heated systems [11].
(3) *Recovery ratio* (*RR*): is the ratio of the amount of water produced per kg of feed. This parameter is also called the extraction efficiency [19]. This is, generally, found to be much lower for the HDH system than conventional systems. The advantage of a low recovery ratio is that complex brine pre-treatment process or brine disposal processes may not be required for this system.
(4) *Energy reuse factor* (f): the ratio of energy recovered from the heated fluid to the energy supplied to the heated fluid [20]. This is another index of heat recovery of the system.

3.1. Closed-air open-water (CAOW) water-heated systems

A typical CAOW system is shown in Fig. 4. The humidifier is irrigated with hot water and the air stream is heated and humidified using the energy from the hot water stream. This process on the psychometric chart is represented by the line 1–2 (Fig. 5). The humidified air is then fed to the dehumidifier and is cooled in a compact heat exchanger using seawater as the coolant. The seawater gets pre-heated in the process and is further heated in a solar collector (Q_{in} indicated in Fig. 4 is the heat absorbed in the solar

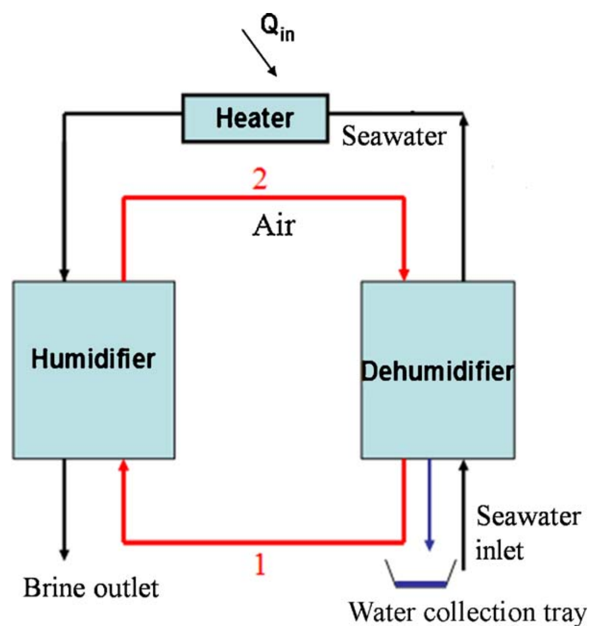


Fig. 4. A typical water-heated CAOW HDH process.

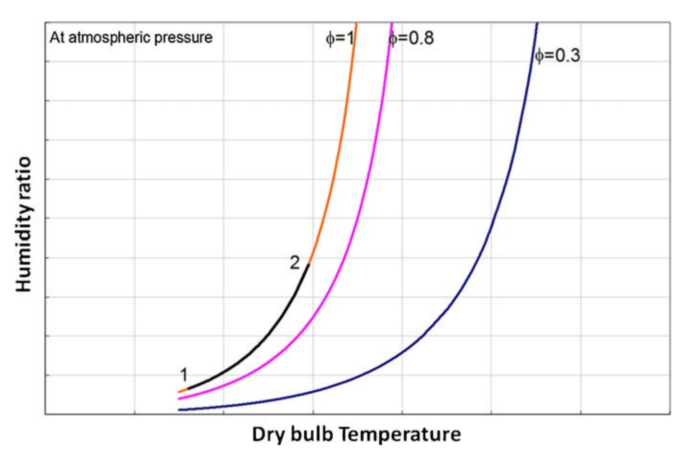


Fig. 5. Water-heated CAOW HDH process on psychometric chart.

collector by the seawater as used in the calculation of GOR) before it irrigates the humidifier. The dehumidified air stream from the dehumidifier is then circulated back to the humidifier. This process on the psychometric chart is represented by the line 2–1 (Fig. 5).

There are several works in the literature on this type of cycle. The important features of the system studied and the main observations from these studies are tabulated in Table 2. Some common conclusions can be drawn from this table. Almost all the investigators have observed that the performance is maximized at a particular value of the water flow rate. There is also an almost unanimous consensus that natural circulation of air yields better efficiency than forced circulation of air for the closed-air water-heated cycle. However, it is not possible to ascertain the exact advantage in performance (for natural circulation) from the data available in literature.

Using the data given in these papers, GOR and specific water production were calculated by the present authors (Fig. 6). The specific water production was found to be between 4 and 12 kg/m^2 day and the GOR varied between 1.2 and 4.5. These values of GOR translate into energy consumption rates from 140 to

Table 2
Features and summary of results from various previous works on water-heated CAOW HDH cycle.

Reference	Unit features	Main observations
Al-Hallaj et al. [80]	• Solar collector (tubeless flat-plate type of 2 m^2 area) has been used to heat the water to 50–70 °C and air is circulated by both natural and forced convection to compare the performance of both of these modes. • Humidifier, a cooling tower with wooden surface, had a surface area 87 m^2/m^3 for the bench unit and 14 m^2/m^3 for the pilot unit. • Condenser area 0.6 m^2 for bench unit and 8 m^2 for the pilot unit.	• The authors noted that results show that the water flow rate has an optimum value at which the performance of the plant peaks. • They found that at low top temperatures forced circulation of air was advantageous and at higher top temperatures natural circulation gives better performance.
Ben-Bacha et al. [21]	• Solar collector used for heating water (6 m^2 area). • There is a water storage tank which runs with a minimum temperature constraint. • Cooling water provided using brackish water from a well. • The packed bed type– used - Thorn trees. • Dehumidifier made of polypropylene plates.	• A daily water production of 19 L was reported. • Without thermal storage 16% more solar collector area was reported to be required to produce the same amount of distillate. • The authors also stated that the water temperature at inlet of humidifier, the air and water flow rate along with the humidifier packing material play a vital role in the performance of the plant.
Farid et al. [67]	• 1.9 m^2 solar collector to heat the water. • Air was in forced circulation. • Wooden shaving packing was used for the humidifier. • Multi-pass shell and tube heat exchanger used for dehumidification.	• 12 L/m^2 production was achieved. • The authors report the effect of air velocity on the production is complicated and cannot be stated simply. • The water flow rate was observed to have an optimum value.
Garg et al. [81]	• System has a thermal storage of 5 L capacity and hence has longer hours of operations. • Solar collector area (used to heat water) is about 2 m^2. • Air moves around due to natural convection only. • The latent is recovered partially.	• The authors conclude that the water temperature at the inlet of the humidifier is very important to the performance of the cycle. • They also observe that the heat loss from the distillation column (containing both the humidifier and the dehumidifier) is important in assessing the performance accurately.
Nafey et al. [82]	• This system is unique in that it uses a dual heating scheme with separate heaters for both air and water. • Humidifier is a packed bed type with canvas as the packing material. • Air cooled dehumidifier is used and hence there is no latent heat recovery in this system.	• The authors reported a maximum production of 1.2 L/h and about 9 L/day. • Higher air mass flow gave less productivity because increasing air flow reduced the inlet temperature to humidifier.
Nawayseh et al. [64]	• Three units constructed in Jordan and Malaysia. Different configurations of condenser and humidifier were studied and mass and heat-transfer coefficients were developed. • Solar collector heats up the water to 70–80 °C. • Air circulated by both natural and forced draft. • Humidifier with vertical/inclined wooden slates packing. • Heat recovered in condenser by pre-heating the feed water.	• The authors observed that the water flow rate has a major effect on the wetting area of the packing. • They also note that natural circulation yields better results than forced circulation. • The heat/mass transfer coefficient calculated were used to simulate performance and the authors report that the water production was up to 5 kg/h.
Müller-Holst et al. [11]	• Closed-air open-water cycle with natural draft circulation for the air. • Thermal storage tank of 2 m^3 size to facilitate 24 h operation. • 38 m^2 collector field size heats water up to 80–90 °C. • Latent heat recovered to heat the water to 75 °C.	• The authors report a GOR of 3–4.5 and daily water production of 500 L for a pilot plant in Tunisia. • There is a 50% reduction in cost of water produced because of the continuous operation provided by the thermal storage device.
Klausner [19,64,65,72]	• A unique HDH cycle with a direct contact packed bed dehumidifier was used in this study. • The system uses waste heat to heat water to 60 °C. • Uses a part of the water produced in the dehumidifier as coolant and recovers the heat from this coolant in a separate heat exchanger.	• The authors demonstrated that this process can yield a fresh water production efficiency of 8% with an energy consumption of 0.56 kWh/kg of fresh water production based on a feed water temperature of only 60° C. • It should be noted that the efficiency is the same as the recovery ratio defined in Section 3.1. • Also the energy consumption does not include the solar energy consumed.
Younis et al. [52]	• 1700 m^2 solar pond (which acts as the heat storage tank) provides heated seawater to be purified. • Forced air circulation. • Latent heat recovered in the condenser to pre-heat seawater going to the humidifier.	• Performance results not reported. • Air flow rate seems to have a major impact on the production of water but surprisingly, water flow rate does not affect the performance.

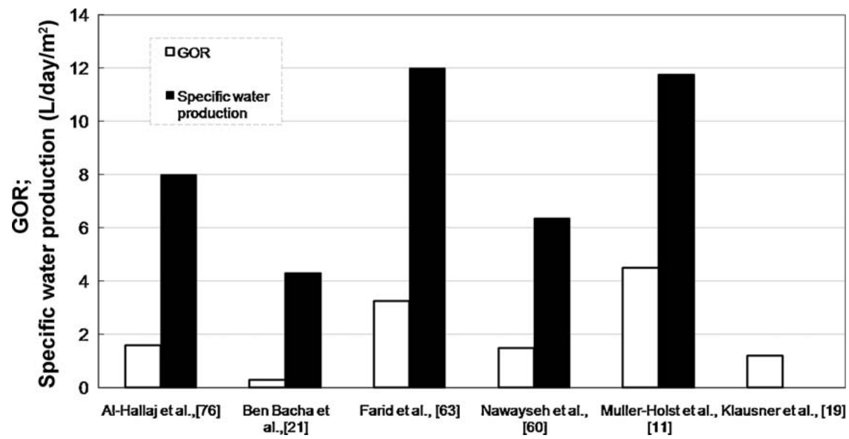


Fig. 6. Performance parameters for various works on water-heated CAOW HDH cycle.

550 kWh/m^3. This is higher than that for conventional technologies like MSF or RO. RO plants, which are the most energy efficient, consume ~4 to 10 kWh/m^3. However, one should keep in mind that the energy supplied is 'free' for these solar HDH systems: GOR for a solar-driven cycle is a measure of thermal performance but it is less directly a measure of water cost.

The low value of GOR achieved by Ben-Bacha et al. [21] was because they did not recover the latent heat of condensation. Instead, they used separate cooling water from a well to dehumidify the air. The higher value of GOR achieved by Müller-Holst et al. [17] was because of high heat recovery. These results tell us the importance of enhanced latent heat recovery to minimize the energy consumption and the cost of CAOW water-heated system. Further, Müller-Holst et al. [17] also reported that the cost can be brought down by ~50% if a heat storage unit is used in the HDH system.

3.2. Multi-effect closed-air open-water (CAOW) water-heated system

To enhance heat recovery, Müller-Holst [11] proposed the concept of multi-effect HDH. Figs. 7 and 8 illustrate an example of this system. Air from the humidifier is extracted at various points and supplied to the dehumidifier at corresponding points. This enables continuous temperature stratification resulting in small temperature gap to keep the process running. This in turn results in a higher heat recovery from the dehumidifier. In fact, most of the energy needed for the humidification process is regained from the dehumidifier bringing down the energy demand to a reported value of 120 kWh/m^3. This system is being commercially manufactured and marketed by a commercial water management company, Tinox GmbH. This is, perhaps, the first instance in which the HDH concept has been commercialized.

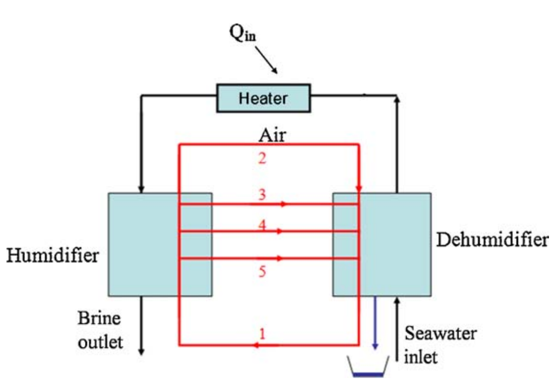


Fig. 7. Multi-effect water-heated CAOW HDH cycle.

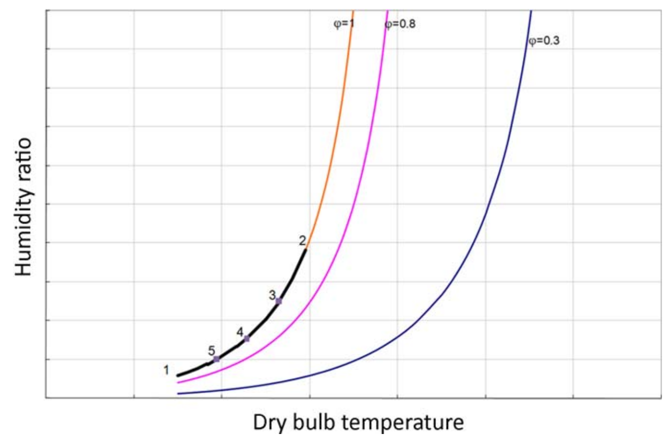


Fig. 8. Multi-effect water-heated CAOW HDH process on psychometric chart.

3.3. Closed-water open-air (CWOA) water-heated systems

A typical CWOA system is shown in Fig. 9. In this system the air is heated and humidified in the humidifier using the hot water from the solar collector and then is dehumidified using outlet water from the humidifier. The water, after being pre-heated in the dehumidifier, enters the solar collector, thus working in a closed loop. The dehumidified air is released to ambient.

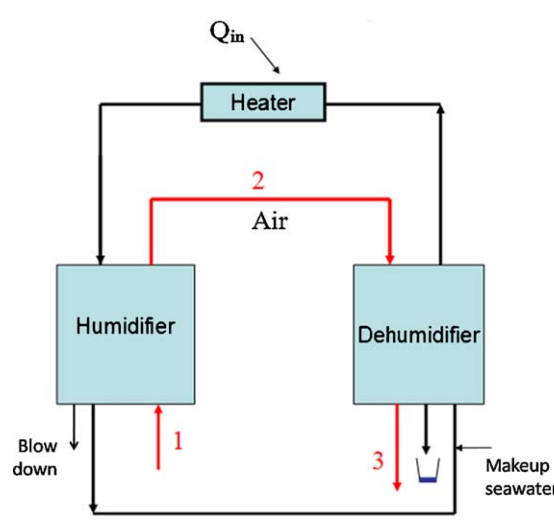


Fig. 9. A typical water-heated CWOA HDH process.

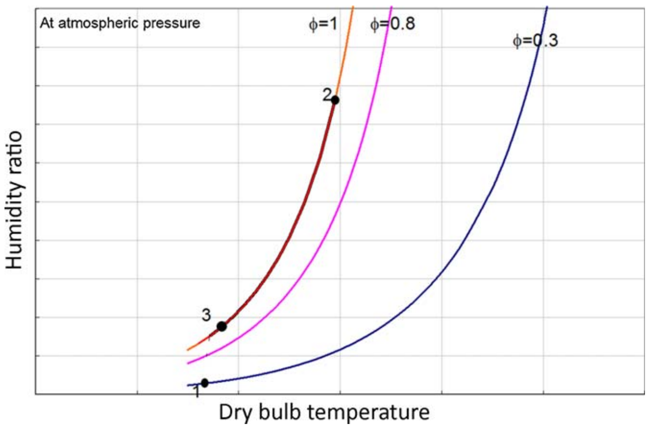


Fig. 10. Water-heated CWOA HDH process on psychometric chart.

The humidification process is shown in the psychometric chart (Fig. 10) by line 1–2. Air entering at ambient conditions is saturated to a point 2 (in the humidifier) and then the saturated air follows a line 2–3 (in the dehumidifier). The air is dehumidified along the saturation line. A relatively small number of works in literature consider this type of cycle. The important features of the system studied and main observations from these studies are shown in Table 3.

One disadvantage of the CWOA is that when the humidification process does not cool the water sufficiently the coolant water temperature to the inlet of the dehumidifier goes up. This limits the dehumidification of the humid air resulting in a reduced water production compared to the open-water cycle. However, when efficient humidifiers at optimal operating conditions are used, the water may be potentially cooled to temperature below the ambient temperature (up to the limit of the ambient wet-bulb temperature). Under those conditions, the closed-water system is more productive than the open-water system.

3.4. Closed-air open-water (CAOW) air-heated systems

Another class of HDH systems which has attracted much interest is the air-heated system (Table 4). These systems are of two types–single and multi-stage systems. Fig. 11 is a schematic diagram of a single stage system. The air is heated in a solar collector to a temperature of 80–90 °C and sent to a humidifier. This heating process is represented by the constant humidity line 1–2 in the psychometric chart (Fig. 12). In the humidifier, the air is cooled and saturated. This process is represented by the line 2–3. It is then dehumidified and cooled in the process 3–1 represented on the saturation line. A major disadvantage of this cycle is that the absolute humidity of air that can be achieved at these temperatures is very low (<6% by weight). This impedes the water productivity of the cycle.

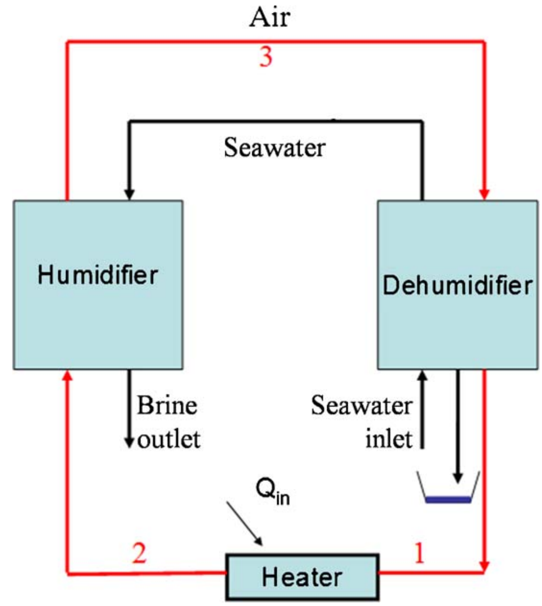


Fig. 11. A typical air-heated CAOW HDH process.

Chafik [18] reported a method to address this problem. He used a multi-stage heating and humidification cycle (Fig. 13). The air after getting heated in the solar collector (line 1–2) and humidified in the evaporator (line 2–3) is fed to another solar collector for further heating (line 3–4) and then to another humidifier (line 4–5) to attain a higher value of absolute humidity. Many such stages can be arranged to attain absolute humidity values of 15% and beyond. Point 2′ in the figure represents the high temperature that has to be reached in a single stage cycle to attain the same humidity as a three-stage cycle. This higher temperature has substantial

Table 3
Previous works on water-heated CWOA HDH cycle.

Reference	Unit features	Brief summary of the paper
Al-Enzi et al. [83]	• Solar collector designed to heat air to 90 °C. • Forced circulation of air. • Cooling water circuit for the condenser. • Heater for preheating water to 35–45 °C. • Plastic packing was used in the humidifier.	• The authors have studied the variation of production in kg/day and heat and mass transfer coefficients with respect to variation in cooling water temperature, hot water supply temperature, air flow rate and water flow rate. • They conclude that the highest production rates are obtained at high hot water temperature, low cooling water temperature, high air flow rate and low hot water flow rate. • The variation in parameters the authors have considered is very limited and hence these conclusions are true only in that range.
Dai et al. [84,85]	• Honeycomb paper used as humidifier packing material. • Forced convention for the air circulation. • Condenser is fin tube type which also helps recover the latent heat by pre-heating seawater.	• It was found that the performance of the system was strongly dependent on the temperature of inlet salt water to the humidifier, the mass flow rate of salt water, and the mass flow rate of the process air. • The authors report that there is an optimal air velocity for a given top temperature of water. • The top water temperature has a strong effect on the production of fresh water.
Khedr [77]	• The system has a packed tower (with 50 mm ceramic Raschig rings) dehumidifier. • The first system to use direct contact condenser in a HDH technology. • The performance parameters are calculated numerically.	• The authors report the GOR for their system as 0.8 which shows that heat recovery is limited. • Based on an economic analysis, they conclude that the HDH Process has significant potential for small capacity desalination plants as low as 10 m^3/day.

Table 4
Previous works on air-heated CAOW HDH cycle.

Reference	Unit features	Brief summary of the paper
Chafik [16,18]	• Solar collectors (four-fold-web-plate, or FFWP, design) of 2.08 m^2 area heat air to 50–80 °C. • Multi-stage system that breaks up the humidification and heating in multiple stages. • Pad humidifier with corrugated cellulose material. • 3 separate heat recovery stages. • Forced circulation of air.	• The author reported that the built system is too costly and the solar air heaters constitute 40% of the total cost. • Also he observed that the system can be further improved by minimizing the pressure drop through the evaporator and the dehumidifiers.
Houcine et al. [15]	• 5 heating and humidification stages. • First two stages are made of 9 FFWP type collector of each 4.98 m^2 area. The other collectors are all classical commercial ones with a 45 m^2 area for the third and fourth stage and 27 m^2 area for the final stage. Air temperature reaches a maximum of 90 °C. • Air is forced-circulated. • All other equipment is the same as used by Chafik [18].	• Maximum production of water was 516 L/day. • Plant tested for a period of 6 months. • Major dilation is reported to have occurred on the polycarbonate solar collectors. • The water production cost for this system is (for a 450–500 L/day production capacity) 28.65 €/m^3 which is high. • ~37% of the cost is that of the solar collector field.
Ben-Amara et al. [53]	• FFWP collectors (with top air temperature of 90 °C) were studied. • Polycarbonate covers and the blackened aluminum strips make up the solar collector. • Aluminum foil and polyurethane for insulation.	• Variation of performance with respect to variation in wind velocity, inlet air temperature and humidity, solar irradiation and air mass flow rate was studied. • Endurance test of the polycarbonate material showed it could not withstand the peak temperatures of summer and it melted. Hence a blower is necessary. • Minimum wind velocity gave maximum collector efficiency.
Orfi et al. [56]	• The experimental setup used in this work uses a solar heater for both air and water (has 2 m^2 collector surface area). There is a heat recovery unit to pre-heat seawater. • The authors have used an evaporator with the heated water wetting the horizontal surface and the capillaries wetting the vertical plates and air moving in from different directions and spongy material used as the packing.	• The authors report that there is an optimum mass flow rate of air to mass flow rate of water that gives the maximum humidification. This ratio varies for different ambient conditions.
Yamali et al. [86]	• A single stage double-pass flat-plate solar collector heats the water. • A pad humidifier is used and the dehumidifier used is a finned tube heat exchanger. • Also a tubular solar water heater was used for some cases. • The authors also used a 0.5 m^3 water storage tank. • Heat is not recovered.	• The plant produced ~4 kg/day maximum. • Increase in air flow rate had no affect on performance. • An increase in mass flow rate of water increased the productivity. • When the solar water heater was turned on the production went up to ~10 kg/day maximum primarily because of the ability to operate it for more time.

Table 5
Standardized efficiencies of certified commercial solar air heaters [45].

Product	Manufacturer	Efficiency
Solarway 6000	Energy Conservation Products and Services	0.122
SunMate Sm-14	Environmental Solar Systems	0.323
MSM-101	SolarMax Heating	0.102
Northern Comfort NC-32	Sunsiaray Solar Inc.	0.266
SolarSheat 1500G	Your Solar Home	0.140

disadvantages for the solar collectors. However, from an energy efficiency point of view, there is not much of an advantage to multi-staging, as the higher water production comes with a higher energy input as compared to single stage systems.

Also, from the various studies in literature summarized in Table 4, we observe that the air-heated systems have higher energy consumption than water-heated systems. This is because air heats up the water in the humidifier and this energy is not subsequently recovered from the water, unlike in the water-heated cycle in which the water stream is cooled in the humidifier.

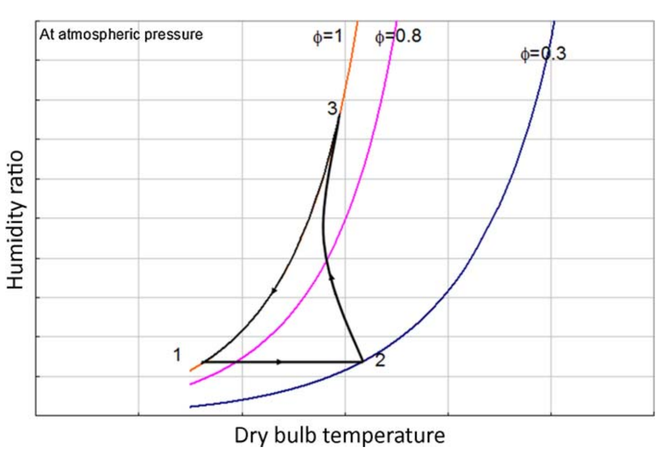


Fig. 12. Air-heated CAOW HDH process on psychometric chart.

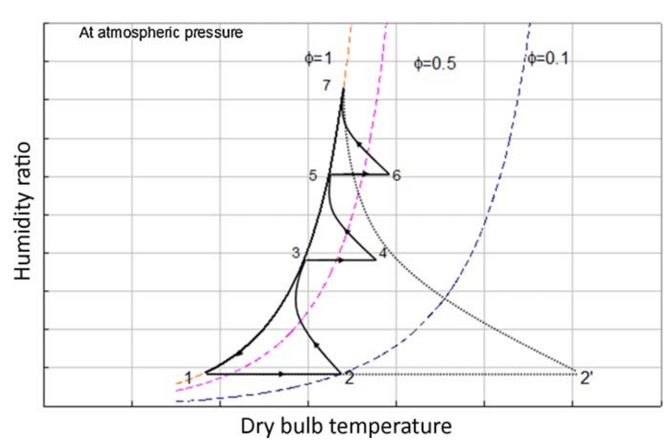


Fig. 13. Multi-stage air-heated CWOA HDH process on psychometric chart.

It should be noted that the CAOW air-heated systems have not been studied so far in literature and hence will not be dealt with in this paper.

4. Review of component designs

4.1. Solar air heater designs

Solar water-heating systems have been studied and used widely for many decades [22–24], and hence extensive knowledge exists on the design of these systems. Solar air heating, in contrast, has not been studied extensively. Considering the importance of solar air heaters to the overall performance and cost of HDH air-heating systems, they are reviewed in the current section. The heaters will be compared on the basis of their collection efficiencies (Eq. (4-1)), which is defined as the useful heat gain of the air stream (in W) divided by the solar irradiation incident on the collector (also in W), unless otherwise noted. This is the same as the instantaneous thermal efficiency test in the ASHRAE 93-2003 Standard [25].

$$\eta = \frac{\text{Heat gained by air}}{\text{Solar incident radiation}} \tag{4-1}$$

The collectors are typically flat plate with large airflow channels. Air flows over or under the absorber plate, and double-pass strategies are sometimes employed. Fig. 14 taken from [26] shows the layout of a typical air-heating collector.

Solar air-heating systems have been used since the World War II for home heating and low temperature applications. The Colorado solar house, built in 1959, utilized a heater that had stacked absorber plates in a panel with a single glazing to achieve a moderate temperature rise for home heating and cooling with 30% collection efficiency [27].

In the 1960s, solar energy was developed in India as a means of cheap energy for crop drying. Gupta and Garg [28] tested several designs that used both corrugated absorber surfaces as well as wire mesh packing over the absorber. This paper also provided an overall efficiency that took into account the power to force air through the heater. It showed that corrugated surfaces performed better than those enhanced with wire mesh, achieving a maximum of 65% overall energy conversion efficiency.

Polymer heaters were also considered. Whillier [29] tested glazing made of Tedlar, a polyvinylfluoride (PVF) film, found that, despite higher heat losses from the Tedlar, its improved transmittance compensated. A glass glazing closer to the absorber plate and Tedlar outer glazing worked better. This new material had the ability to increase efficiency and was also resistant to corrosion. Later Bansal [30] reported on two designs built in 1982 and tested in the environment over a long period of time. PVF offered better thermal performance than PVC. Both materials were subject to UV degradation, which resulted in shorter lifetimes, but they offered significant cost savings over glass and metals. McCullough et al. [31] also made use of PVF materials, as well as polycarbonate (PC) in place of glass.

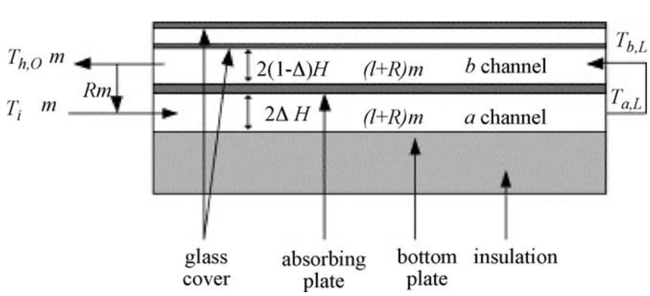


Fig. 14. Flow through solar air collector [26].

Beginning with the 1973 oil crisis, more research was done on alternative energy, including solar air heaters. Satcunanathan and Deonarine [32] tested the use of multiple glazings and of passing air between the glazings. The air passed under a corrugated absorber (parallel to the ribs) to be heated. Efficiency gains of 10–15% were found when air was passed between the two glazings, as it kept the outer glazing cooler and reduced convective losses. The vast majority of solar air heaters were also patented after this period. Many of these designs took up the issue of poor heat transfer associated with a laminar flow over a smooth absorber plate. Designs by Severson [33] and Schmidt [34] used a perforated plate to create jets of air that pointed at the absorber. These designs have difficulties with large pressure drops and low overall efficiencies. Another design by Vincent [35] shapes the collector into a dome with a double glazing and air circulated on top of the absorber and under the second glazing. This design maximizes exposure to the sun, but also suffers greater losses because of its large area, parts of which may not be exposed to the sun and thus only adding to losses.

In the 1990s research continued, particularly in India where solar energy was being used for inexpensive low-grade heat. Choudhury and Garg [36] compared designs that used packing materials above and below the absorber plate while passing air through the packing. Efficiencies up to 70% were obtained when the air and packing were above the absorber plate. The study also compared different packing materials and found low porosity and small particle diameter packing worked best. Sharma et al. [37] evaluated a wire matrix air heater, where air was flowing through the matrix. The heater was intended for a crop drying application.

Modern air heater designs have focused mainly on improving convective heat transfer at the absorber. Mittal and Varshney [38] investigated using wire mesh as a packing material, with air flowing between the absorber and the second glazing through the mesh, achieving a collector efficiency of 70%. Mohamad [39] found that a packed bed of porous media improved heat transfer as well as pre-warming the air by first running it between two glazing plates. This also improved collector efficiency by reducing heat loss to the environment, and helped achieve an overall efficiency, which accounts for pumping losses for moving air through the collector of 75%. Esen [40] compared several obstacles mounted on a flat plate to a plain flat plate and found that short triangular shaped barriers improved heat-transfer efficiency the most by breaking up the boundary layer and reducing dead zones in the collector. Romdhane [41] used small extensions from a metal plate to improve mixing of air on the plate. These extensions had the advantage of not increasing pressure drop like packed bed solar air heaters. Ho [26] increased the collector efficiency of a flat metal absorber plate to 68% by running the air above and below the absorber plate. The flow turns 180° to move back above the plate. This configuration increases pressure drop in the flow, but the paper does not specify how much. Ramadan et al. [42] also reported an efficiency increase using double-pass heating.

Other attempts have been made to improve existing flat-plate absorber with limited success. These designs sacrifice efficiency for simplicity. Koyuncu [43] compared several flat-plate designs, with one ribbed plate design, and several glazing configurations. The most efficient, at 45.8%, was flat black metal plate with a single polymer glazing, and air passing over the absorber. Matrawy [44] used fins below the absorber plate to enhance heat transfer to the air as it flowed under the absorber, but only achieved 50% collector efficiency.

To date there are no commercial systems that utilize solar air heaters for solar desalination, only for home heating and crop drying. Most products have moderate temperature rise, and are very expensive. Several of these products were rated by the Solar Collector and Certification Corporation, which gives efficiency data

versus temperature rise normalized to solar radiation. Table 5 shows the efficiency of the commercial collectors rated by [45] evaluated at a solar irradiation of 1 kW/m^2 and a temperature rise of 50 °C, which are representative values for a HDH desalination application. The best performing collector under these conditions is the SunMate Sm-14, achieving only 32% efficiency.

4.1.1. Standardized comparison of designs

As with other heat exchangers, a solar air heater decreases in efficiency with a greater temperature rise due to increased loss. The most common way of showing solar air heater efficiency is to plot the efficiency versus the normalized heat gain, which is the rise in air temperature divided by the solar irradiation flux. The normalized gain will decrease with increasing air mass flow rate. Heaters that have a given loss coefficient will be less efficient at higher normalized gain. Fig. 15a shows the reported efficiencies of solar air heaters in the literature as a function of normalized heat gain. The high efficiency commercial solar collector, the SunMate Sm-14, is included for comparison. Those heaters with lower losses and thus better performance will have a higher efficiency at a high normalized gain. Therefore the better preforming heaters are found in the top left of the graph.

The top performing heaters [32,38,42,46,47] are highlighted in light grey. The two outliers [39,41] do not follow the trend in the other data and were not included in the comparison.

When considering the design of a solar air heater, Duffie and Beckman [49] suggest two design parameters that vary based on collector design. One is the overall heat loss coefficient, U_L, which is related to the heat-transfer coefficients in the collector, and which needs to be minimized. This is given by Eq. (4-2) for a flat-plate air heater with air flowing over the absorber.

$$U_L = \frac{(U_b + U_t)(h_1 h_2 + h_1 h_r + h_2 h_r) + U_b U_t (h_1 + h_2)}{h_1 h_r + h_2 U_t + h_2 h_r + h_1 h_2} \tag{4-2}$$

The second parameter is F', which is the useful heat gain coefficient or the ratio of actual energy gain to the energy gain that would result if the absorber plate was at the local fluid temperature. This ratio needs to be maximized to enhance efficiency. Eq. (4-3) gives F' for the same flat-plate air heater.

$$F' = \frac{h_1 h_r + h_2 U_t + h_2 h_r + h_1 h_2}{(U_t + h_r + h_1)(U_b + h_2 + h_r) - {h_r}^2} \tag{4-3}$$

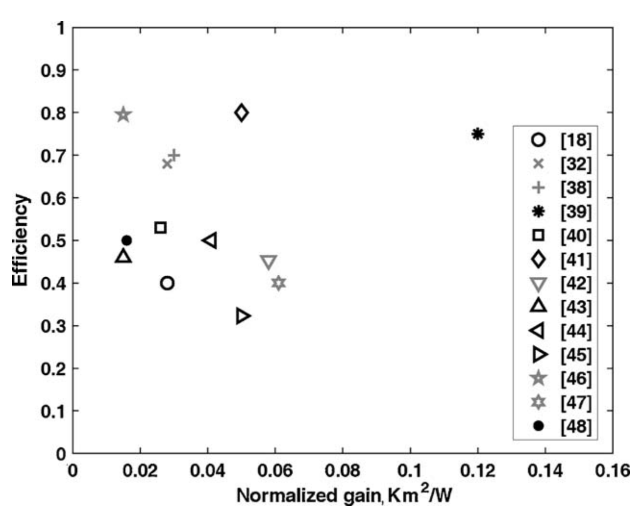


Fig. 15. Performance comparison of solar air heaters.

To see how each parameter fits into the overall useful heat gain Eq. (4-4), or the overall collector governing equation, is also given.

$$q''_u = F'(S - U_L(T_f - T_a)) \tag{4-4}$$

S is the total energy that is absorbed by the absorber. U_b and U_t are the overall heat-transfer coefficients from the top and bottom of the air stream to the outside, respectively, h_1 is the heat-transfer coefficient from the glazing plate to the air stream, h_2 is the heat-transfer coefficient from the absorber to the air stream, and h_r is the linearized radiation heat-transfer coefficient from the absorber to the glazing.

4.2. Humidifier designs

Many devices are used for air humidification including spray towers, bubble columns, wetted-wall towers and packed bed towers [50]. The principle of operation for all of these devices is same. When water is brought into contact with air that is not saturated with water vapor, water diffuses into air and raises the humidity of the air. The driving force for this diffusion process is the concentration difference between the water–air interface and the water vapor in air. This concentration difference depends on the vapor pressure at the gas–liquid interface and the partial pressure of water vapor in the air.

Any of the above mentioned devices can be used as a humidifier in the HDH system. A spray tower for instance consists essentially of a cylindrical vessel in which water is sprayed at the top of the vessel and moves downward by gravity dispersed in droplets within a continuous air stream flowing upward. These towers are simple in design and have minimal pressure drop on the gas side. However, there is a considerable pressure drop on the water side due to the spray nozzles. Also, mist eliminators are always necessary due to the tendency of water entrainment by the air leaving the tower. It is generally known that this device has high capacity but low efficiency. The low efficiency is as a result of the low water hold-up due to the loose packing flow [51]. The diameter-to-length ratio is a very important parameter in spray tower design. For a large ratio air will be thoroughly mixed with the spray. Small diameter-to-length ratio will let the spray quickly reach the tower walls, forming a film becoming ineffective as a spray. Design of spray towers requires knowledge of heat and mass transfer coefficients as well as the contact surface area of the water droplets. Many empirical correlations and design procedures are given in Kreith and Boehm [51].

Younis et al. [52] and Ben-Amara et al. [53] used a spray tower as the humidifier in their HDH systems. Ben-Amara et al. [53] tested the spray tower humidifier by varying the ratio of water-to-dry air mass flow rate and keeping the inlet water temperature and absolute humidity constant. The inlet air temperature (80 °C) was higher than the water spray temperature (60 °C). They found that increasing the amount of water sprayed increased the absolute outlet humidity. However, further increase in the water quantity resulted in air cooling and this condensed some of the water vapor content in the air. This means a decrease in the absolute humidity, although the outlet air is always saturated. Therefore, for air-heated HDH cycles there is an optimum value of the mass flow ratio which gives maximum air humidity. This fact promotes the use of multi-stage air heater and humidifier combinations to increase the fresh water production.

Exactly opposite in principle to the spray tower is the bubble column. In the bubble column, a vessel is filled with water and air bubbles are ejected from several orifices located at the bottom of the vessel. Water diffuses into the air bubbles and causes the outlet air to be humidified. These columns are simple in design; however, the diffusion of water into the air bubbles depends on many

parameters such as bubble diameter, bubble velocity, gas hold-up (the ratio of air bubbles-to-water volume), water and air temperatures as well as the heat and mass transfer coefficients. In HDH desalination systems, bubble columns have not been used as humidifiers so far. However, El-Agouz and Abugderah [54] investigated experimentally the performance of a single stage bubble column using air bubbles passing through seawater. They studied the influence of operating conditions on the vapor content difference and the humidification efficiency, which showed strong dependence on saline water temperature and the air velocity. Moreover, the inlet air temperature has a small effect on the vapor content difference. The maximum experimentally obtained vapor content difference of the air was 222 g/kg of dry air at 75 °C of water and air temperatures. However, other geometrical factors such as the orifice diameter, number of orifices, water head height and column diameter were not considered. It is important to mention that there are many empirical correlations for these parameters in Treybal [50] and Lydersen [55]. Therefore an optimum design and performance evaluation study can be carried out before using the bubble columns in HDH systems.

Wetted-wall towers have been used as humidifier in HDH systems by Müller-Holst et al. [17] and Orfi et al. [56]. In a wetted-wall tower, a thin film of water is formed running downward inside a vertical pipe, with air flowing either co-currently or counter-currently. Water is loaded into the top of the tower and a weir distributes the flow of water around the inner perimeter of the tube that wets the inner surface of the tube down its length. Such devices have been used for theoretical studies of mass transfer, since the contact area can be calculated, accurately. In Müller-Holst et al.'s system [17], heated water was distributed onto vertically hanging fleeces made of polypropylene and trickled downwards. The air move in countercurrent flow to the brine through the humidifier and becomes saturated at the outlet. On the other hand, Orfi et al. [56] used a different design for their wetted-wall humidifier. To improve the heat and mass exchange process, they covered the wooden vertical wetted-walls with a cotton wick to reduce the water flowing velocity and use the capillary effect to keep the vertical walls always wetted. Their design shows higher performance with about 100% humidification efficiency.

To increase the humidification efficiency, packing is typically used. This helps by increasing the dispersion of water droplets, the contact area and contact time. Devices that contain packing material are known as packed bed towers and special types that are used to cool water are called cooling towers. These are vertical columns filled with packing materials with water sprayed at the top and air flows in counter or cross-flow arrangement. Packed bed towers have been used by many researchers as a humidifier device in HDH desalination systems because of the higher effectiveness. Different packing materials have been used as well (Table 6). The factors influencing the choice of a packing are its heat and mass transfer performance, the quality of water, pressure drop, cost and durability. Over the last 30 years, there has been a gradual change in the types of fill used in packed bed towers as indicated by Wallis and Aull [57]. The most dramatic change has been the introduction of film fills that provide significantly higher thermal performance through the increase of water-to-air contact area and a reduction in pressure drop. However, in HDH desalination application, due to high fouling potential, these benefits are forfeited and the older splash-type fill packing is used. Mirsky and Bauthier [58] presented a history of the development of packing materials while Aull and Krell [59] investigated the performance of various film-type fills. The Merkel, Poppe and epsilon-NTU heat and mass transfer methods of analysis are the cornerstone of cooling tower performance evaluation. A critical evaluation and refinement of these methods is given by Kloppers [60] in his PhD dissertation while design and performance evaluation of cooling towers are discussed in somewhat detail by Kroger [61].

Table 6
Packing material used in packed bed towers for HDH systems.

Author	Packing material
Khedr [77]	Ceramic Raschig rings
Farid et al. [67]	Wooden shaving
Al-Hallaj et al. [80]	Wooden surface
Nawayseh et al. [64]	Wooden slates packing
Dai and Zhang [84,85]	Honeycomb paper
Garg et al. [81]	Indigenous structure
Ben-Bacha et al. [21]	Thorn trees
Efat Chafik [18]	Corrugated cellulose material
Nafey et al. [82]	Canvas
Klausner [17]	HD Q-PAC
Al-Enzi et al. [83]	Plastic packing
Houcine et al. [15]	Corrugated cellulose material
Yamali et al. [86]	Plastic packing

To evaluate the performance of an air humidifier, an efficiency or effectiveness should be used. Many researchers defined humidifier efficiency as [18,54,56],

$$\eta = \frac{\omega_{\text{out}} - \omega_{\text{in}}}{\omega_{\text{out,sat}} - \omega_{\text{in}}}$$

where ω_{out} is outlet humidity ratio; ω_{in} is inlet humidity ratio; and $\omega_{\text{out,sat}}$ is outlet humidity ratio at saturation.

The maximum humidity difference in this definition assumes that the outlet air is saturated at the exit air temperature. This definition is basically used for evaporative coolers [62] where unsaturated air passes through a packing material wetted with water that is sprayed at the top of the packing. The sprayed water is circulated and at steady state condition its temperature reaches the wet-bulb temperature of the inlet air. In this case the air temperature decreases and it approaches the wet-bulb temperature. This humidifier efficiency cannot be used if the inlet air is saturated because there will be no humidity increase. However, if the inlet water temperature is higher than the air temperature or steam is injected into air stream, the air in this case will be heated and humidified. In this case also, the air will be near the saturation condition, thus the efficiency definition described above will not represent how efficient is the humidification process.

4.3. Dehumidifiers

The types of heat exchangers used as dehumidifiers for HDH applications vary. For example, flat-plate heat exchangers were used by Müller-Holst et al. [17]. Others used finned-tube heat exchangers [16,18,54]. A long tube with longitudinal fins was used in one study [63], while a stack of plates with copper tubes mounted on them in another study [64,65] used a horizontal falling film-type condenser. Direct contact heat exchangers were also used as a condenser in some other studies [66] in combination with a shell- and-tube heat exchanger to provide enhanced condensation and improved heat recovery for the cycle.

A flat-plate heat exchanger made of double webbed slabs of propylene was used by Müller-Holst et al. [17] in his HDH system. The distillate runs down the plates trickling into the collecting basin. Heat recovery is achieved by transferring heat to the cold seawater flowing inside the flat-plate heat exchanger. The temperature of seawater in the condenser increases from 40 to 75 °C. In a similar study, Chafik [16,18] used seawater as a coolant wherein the water is heated by the humid air before it is pumped to the humidifiers. Three heat exchangers were used in three different condensation stages. An additional heat exchanger is added at the intake of sea water (low temperature level) for further dehumidification of air. The heat exchangers (or dehumidifiers) are finned-tube type air coolers. They developed a theoretical model

by using TRNSYS to calculate heat-transfer coefficients from both the hot- and cold-sides of the heat exchanger from which the system operating conditions were set. It is important to note that to withstand corrosive nature of seawater; stainless steel is used for frames, collecting plates, while the fins are made of aluminum. In addition, special attention was exercised to avoid leakage of distillate water.

Different designs of condensers in a HDH cycle were used by Farid et al. [63,67]. In a pilot plant built in Malaysia, the dehumidifier was made of a long copper-galvanized steel tube (3 m length, 170 mm diameter) with 10 longitudinal fins of 50 mm height on the outer tube surface and 9 fins on the inner side. In another location, they used a simplified stack of flat condenser made of 2 m $\times$ 1 m galvanized steel plates with long copper tubes mounted on each side of the plate to provide a large surface area. The condenser size was made large, particularly to overcome the small heat-transfer coefficients both on the air- and water-sides due to relatively low air velocity, as well as low water flow rates.

In another design, the dehumidifier was made of 27 m long copper pipe having a 10 mm OD, mechanically bent to form a 4 m long helical coil fixed in the PVC pipe. The pre-heated feed water was further heated in a flat-plate collector. The hot water leaving the collector was uniformly distributed over a wooden shaving packing in a 2 m long humidifier. It is important to note that the condenser or dehumidifier was made of hard PVC pipes connected to form a loop with the blower fixed at the bottom. The condenser was made of a copper pipe mechanically bent to form a helical coil fixed in the PVC pipe.

Two types of condensers were reported in another study by [64]. These were constructed from galvanized steel plates for both the bench and pilot units. In the pilot unit, a copper tube having 11 mm OD and 18 m long was welded to the galvanized plate in a helical shape. The tube outside diameter and length in the bench unit were 8 mm and 3 m, respectively. Either one or two condensers, connected in series, were fixed vertically in one of the ducts for both the units. In one unit, the condenser was simply a 3 m long cylinder having a diameter of 170 mm and made of galvanized steel plates. Ten longitudinal fins were soldered to the outer surface of the cylinder and nine similar fins were soldered to the inner surface. The height of inside and outside fins was 50 mm. The thickness of the plate that was used to make the cylinder and the fins was 1.0 mm. A copper tube having 9.5 mm inside diameter was soldered to the surface of the cylinder. The condenser was fixed vertically in the 316 mm diameter PVC pipe which is connected to the humidifier section by two short horizontal pipes.

Bourouni et al. [65] used a condenser made of polypropylene which was designed to work at low temperatures (70–90 °C) for a HDH system. It is similar to a horizontal falling film-type condenser. At the top of the dehumidifier, the hot humid air is forced down where the distilled water is recovered. It is important to note that heat recovery in an HDH system requires a larger heat-transfer area for improving the overall system performance. For this reason, 2000 m of tubes are used in the evaporator, while 3000 m of tubes in the condenser.

The system that Orfi et al. [56] used had two solar heaters, one for heating water and the other for heating air. The condenser, that uses seawater for cooling, consists of a chamber with a rectangular cross-section. It contains two rows of long cylinders made of copper in which the feed water flows. Longitudinal fins were soldered to the outer surface of the cylinders. The condenser is characterized by heat-transfer surface area of 1.5 m^2 having 28 m as a total length of the coil.

Packed bed direct contact heat exchangers were used in a few researchers [56,68,69], because the film condensation heat transfer is tremendously degraded in the presence of non-condensable gas. An additional shell and tube heat exchanger is used to cool the desalinated water from which a portion is re-circulated and sprayed in the condenser.

Threlkeld [70] explains the governing equations for the dehumidifier in a differential form. Also, design correlations for both friction factor and heat-transfer coefficients that can be used for dehumidifiers are summarized by Pacheco-Vega et al. [71].

The standard method as developed by McQuiston [72,73] considers finned-tube multi-row multi-column compact heat exchangers and predicts heat and mass transfer rates using Colburn j-factors along with flow rate, dry- and wet-bulb temperatures, fin spacing and other dimensions. The air side heat-transfer coefficient is based on log-mean temperature difference for the dry surface whereas under the condensing conditions, the moist air enthalpy difference is used as a driving potential.

Pacheco-Vega et al. [71] used neural network techniques and the experimental data collected by McQuiston, to create a trained network that predicted the exchanger's heat rate directly. Remarkably accurate results were obtained as compared with the method of using correlations of heat and mass transfer coefficient and Colburn j-factors. They focused on the exchanger heat rate since it is the value ultimately desired by users. A significant improvement in the accuracy of predictions compared to the conventional j-factor approach was demonstrated, e.g., 56.9% less error for drop wise condensation and 58.6% less error for film wise condensation have been reported.

5. Alternate cycles resembling the HDH process

5.1. Dew-Vaporation technique

Beckmann has invented [74] and investigated [75] a desalination technology that works on the humidification–dehumidification principle. They call it the 'Dew-Vaporation' technique (Fig. 16). Unlike the HDH process, it uses a common heat-transfer wall between the humidifier (which they call the evaporation chamber) and the dehumidifier (which they call the dew-formation chamber). The latent heat of condensation is directly recovered through this wall for the humidification process. It is reported that the use of this common heat-transfer wall makes the process energy efficient.

In this process the saline water, after being pre-heated using the exit distillate water stream, wets the heat-transfer wall and is heated by means of the latent heat of condensation from the

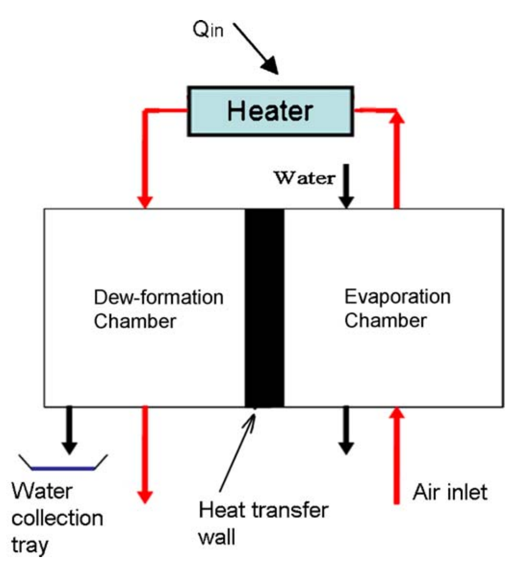


Fig. 16. Dew-Vaporation process.

dew-formation chamber. It then evaporates into the air stream, humidifying it. The humidified air stream is then heated using an external source and is fed to the dehumidifier at a temperature higher than the temperature of air leaving the humidifier.

While, heat is directly recovered from the dew-formation tower, it should be noted that the condensation process itself is relatively ineffective. The dehumidified air exits the tower at a high temperature of around ~50 °C (compared to 30–35 °C in a HDH cycle). Also, the coupling of the humidification and dehumidification processes sacrifices the modularity of the HDH system and the related opportunities to optimize subsystem design and performance separately. However, despite these possible drawbacks this technology appears to have some potential. Hamieh et al. [20] have reported very high values of heat recovery from this system.

5.2. Diffusion-driven desalination technique

Investigators at University of Florida have patented [76] an alternate desalination process that works on the HDH principle. They call it the 'diffusion-driven desalination' (DDD) process. The system is similar to the closed-air open-water HDH cycle, but it uses a direct contact dehumidifier in place of the non-contact heat exchanger normally used for condensation in the HDH systems. The dehumidification process uses a portion of the distilled water produced from the cycle as a coolant. A chiller is used to provide the distilled water at a low temperature. In a similar system, Khedr [77] had earlier proposed an HDH system with a direct contact dehumidifier having ceramic Raschig rings as the packing material. The specific energy demand of the DDD process (GOR ~1.2) is higher for this cycle than for a normal HDH cycle in which the latent heat in the dehumidifier is not recovered.

5.3. Atmospheric water vapor processors

Wahlgren [78] reviewed various processes that extract the humidity from ambient air. These processes are called dew collection processes and the system is sometimes called an atmospheric vapor processor. Three different methods have been applied in these systems: (1) surface cooling using heat pumps or radiative cooling devices; (2) using of solid/liquid desiccants to concentrate the moisture in atmospheric air before condensing it out; and (3) convention-induced dehumidification.

While it may seem promising to take advantage of air that is already humidified and a cycle which consists of only dehumidification (which is by itself exothermic), some major drawbacks accompany this concept of water extraction. The absolute humidity in ambient air found in most places around the world is low, and hence to produce a reasonable amount of water a large amount of air needs to circulate through the process equipment. Also, even though the dehumidification process is exothermic the possibility of extracting any thermodynamic advantage from it exists only when a lower temperature sink is available.

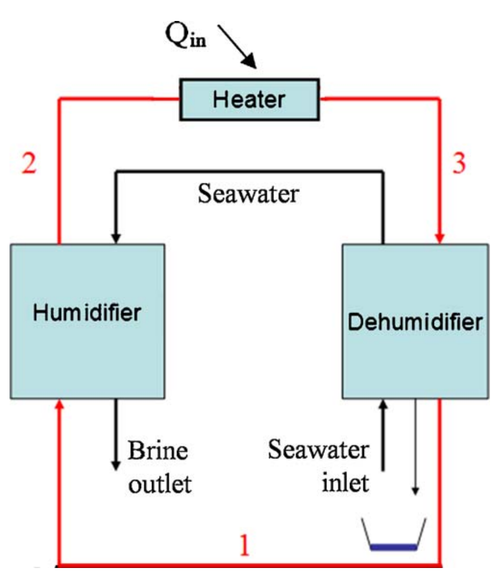


Fig. 17. A modified air-heated CAOW HDH process.

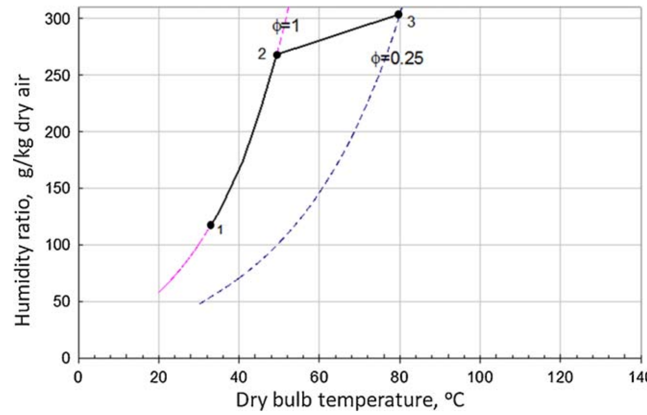


Fig. 18. Psychometric representation of the proposed process in an enthalpy versus dry bulb temperature chart.

6. Possible improvements to the HDH cycle

We observe that most studies in the literature consider cycles that heat the air before the humidifier (in single- or multi-stage), which causes the heat recovery to be reduced since the air gets cooled in the humidifier. If the heater is placed after the humidifier (Fig. 17), saturated air from the humidifier is heated and sent to the dehumidifier. Seawater gets heavily pre-heated in the dehumidifier and the air in turn is heated and humidified in the humidifier [79].

There are two advantages to this cycle: (1) the condensation process occurs in a higher temperature range than the evaporation process, and hence heat is recovered efficiently; and (2) the enthalpy curves for humid air are such that a large temperature rise can be achieved easily for this cycle. This can be observed from the enthalpy-temperature diagram shown in Fig. 18. Even for water-heated cycles, the humidification process occurs at higher air temperatures than the dehumidification process and the heat recovery is affected by that as well. Thus, the proposed cycle should have better heat recovery than all the systems presented in the literature.

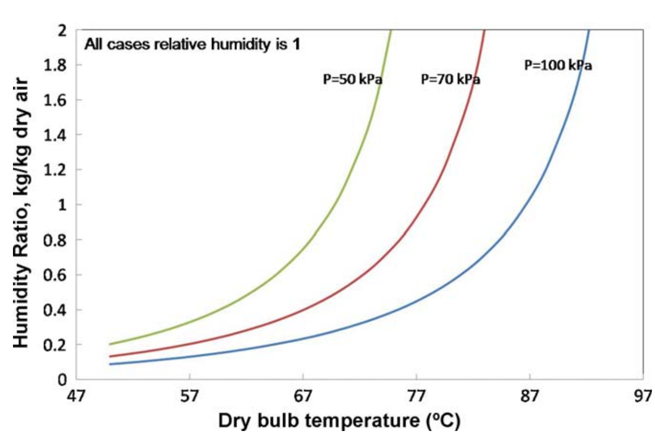


Fig. 19. Effect of pressure on water carrying capability of air.

It can be observed that all the HDH systems in literature operate at atmospheric pressures only. The humidity ratios are much higher at pressures lower than atmospheric pressure [87–89]. This is expected to increase the water production many times for the HDH cycle [79]. For example, at a dry bulb temperature of 60 °C, the humidity ratio at 50 kPa is ~150% higher than at atmospheric pressure (Fig. 19). Last but not least, accurate seawater properties (e.g. [90]) should be used for thermodynamic analysis of HDH cycles. As the differences between fresh water and seawater thermophysical properties may lead to errors in the performance evaluation of the cycle under investigation [90].

7. Conclusions

Solar humidification–dehumidification desalination technology has been reviewed in detail in this paper. From the present review it is found that among all HDH systems, the multi-effect CAOW water-heated system is the most energy efficient. For this system, the cost of water production is ~US$ 3–7/m^3 [17]. Even though this is higher than that for RO systems working at similarly small capacities (5–100 m^3/day), the HDH system has other advantages for small-scale decentralized water production. These advantages include much simpler brine pre-treatment and disposal requirements and simplified operation and maintenance. Methods to further improve the performance of the HDH cycle have also been proposed in this paper. These methods include sub-atmospheric operation. Further research needs to be carried out to realize the full potential of these ideas and the HDH concept in general.

Acknowledgements

The authors would like to thank the King Fahd University of Petroleum and Minerals for funding the research reported in this paper through the Center for Clean Water and Clean Energy at MIT and KFUPM.

References

[1] United Nations. The millenium development goals report. New York: United Nations; 2008.
[2] El-Dessouky HT, Ettouney HM. Fundamentals of salt water desalination. Elsevier; 2002.
[3] Wilf. The guidebook to membrane desalination technology. L'Aquila, Italy: Balaban Desalination Publications; 2007.
[4] Strathmann H. Ion-exchange membrane separation processes. New York: Elsevier; 2004.
[5] Alshareff FF. Investment opportunities in the desalination sector of the Kingdom of Saudi Arabia resulting from privatization and restructuring. In: Saudi water and power forum; 2008.
[6] Sauvet-Goichon B. Ashkelon desalination plant—a successful challenge. Desalination 2007;203:75–81.
[7] Cath TY, Childress AE, Elimelech M. Forward osmosis: principles, applications, and recent developments. Journal of Membrane Science 2006;281:70–87.
[8] Qiblawey HM, Banat F. Solar thermal desalination technologies. Desalination 2008;220:633–44.
[9] Trieb F, Kabariti M, Shahin W, Taher A, Altowaie H, Sufian T, et al. Concentrating solar power for seawater desalination. Final report. Stuttgart: German Aerospace Center (DLR); 2007.
[10] Peter-Varbanets M, Zurbru C, Swartz C, Pronk W. Decentralized systems for potable water and the potential of membrane technology. Water Research 2009;43:245–65.
[11] Müller-Holst H. Solar thermal desalination using the multiple effect humidification (MEH) method. In: Book chapter, solar desalination for the 21st century; 2007. pp. 215–25.
[12] Tiwari GN, Singh HN, Tripathi R. Present status of solar distillation. Solar Energy 2003;75(5):367–73.
[13] Fath HES. Solar distillation: a promising alternative for water provision with free energy, simple technology and a clean environment. Desalination 1998;116:45–56.
[14] Lawand TA. Systems for solar distillation. Brace Research Institute; 1975 [Report no. R 115].
[15] Houcine I, Amara MB, Guizani A, Maalej M. Pilot plant testing of a new solar desalination process by a multiple-effect-humidification technique. Desalination 2006;196:105–24.
[16] Chafik E. Design of plants for solar desalination using the multi-stage heating/humidifying technique. Desalination 2004;168:55–71.
[17] Müller-Holst H, Engelhardt M, Herve M, Scholkopf W. Solar thermal seawater desalination systems for decentralized use. Renewable Energy 1998;14(1–4):311–8.
[18] Chafik E. A new type of seawater desalination plants using solar energy. Desalination 2003;156:333–48.
[19] Klausner JF, Mei R, Li, Y. Innovative fresh water production process for fossil fuel plants. U.S. DOE—Energy Information Administration annual report; 2003.
[20] Hamieh BM, Beckmann JR. Seawater desalination using Dew-vaporation technique: theoretical development and design evolution. Desalination 2006;195:1–13.
[21] Ben-Bacha H, Damak T, Bouzguenda M. Experimental validation of the distillation module of a desalination station using the SMCEC principle. Renewable Energy 2003;28:2335–54.
[22] Garg HP. Year round performance studies on a built-in storage type solar water heater at Jodhpur, India. Solar Energy 1975;17:167–72.
[23] Garg HP. Solar water heating systems. In: Proceedings of the workshop on solar water heating systems; 1985.
[24] Eggers-Lura A. Solar energy for domestic heating and cooling: a bibliography with abstracts, and a Survey of literature and information sources. Pergamon Press; 1978.
[25] Rojas D, Beermann J, Klein SA, Reindl DT. Thermal performance testing of flat-plate collectors. Solar Energy 2008;82(8):746–57.
[26] Ho CD, Yeh HM, Wang RC. Heat-transfer enhancement in double-pass flat-plate solar air heaters with recycle. Energy 2005;30(15):2796–817.
[27] Lof GOG, El-Wakil MM, Chiou JP. Residential heating with solar-heated air—Colorado solar house. ASHRAE Journal 1963;5(10):77–86.
[28] Gupta CL, Garg HP. Performance studies on solar air heaters. Solar Energy 1967;11(1):25–31.
[29] Whillier A. Plastic covers for solar collectors. Solar Energy 1963;7(3):148–51.
[30] Bansal NK. Thermal performance of plastic film solar air and water heaters. International Journal of Energy Research 1987;11(1):35–43.
[31] McCullough RW. Solar air heater. US Patent 4,262,657 (1977).
[32] Satcunanathan S, Deonarine S. A two-pass solar air heater. Solar Energy 1973;15(1):41–9.
[33] Severson AM. Solar air heater. US Patent 4,085,730 (1978).
[34] Schmidt RN. Solar air heater. US Patent 4,085,729 (1976).
[35] Vincent OW. Dome solar air heater. US Patent 4,236,507 (1977).
[36] Choudhury C, Garg HP. Performance of air-heating collectors with packed airflow passage. Solar Energy 1993;50(3):205–21.
[37] Sharma VK, Sharma S, Mahajan RB, Garg HP. Evaluation of a matrix solar air heater. Energy Conversion and Management 1990;30(1):1–8.
[38] Mittal MK, Varshney L. Optimal thermo hydraulic performance of a wire mesh packed solar air heater. Solar Energy 2006;80(9):1112–20.
[39] Mohamad AA. High efficiency solar air heater. Solar Energy 1997;60(2):71–6.
[40] Esen H. Experimental energy and exergy analysis of a double-flow solar air heater having different obstacles on absorber plates. Building and Environment 2008;43(6):1046–54.
[41] Romdhane BS. The air solar collectors: comparative study, introduction of baffles to favor the heat transfer. Solar Energy 2007;81(1):139–49.
[42] Ramadan MRI, El-Sebaii, Aboul-Enein S, El-Bialy E. Thermal performance of a packed bed double-pass solar air heater. Energy 2007;32(8):1524.
[43] Koyuncu T. Performance of various designs of solar air heaters for crop drying applications. Renewable Energy 2006;31(7):1073–88.
[44] Matrawy KK. Theoretical analysis for an air heater with a box-type absorber. Solar Energy 1998;63(3):191–8.
[45] Solar Rating and Certication Corporation, Directory of certied solar collector ratings, www.solar-rating.org.
[46] Sahu M, Bhagoria J. Augmentation of heat transfer coefficient by using 90 degree broken transverse ribs on absorber plate of solar air heater. Renewable Energy 2005;30(13):2057–73.
[47] Close DJ. Solar air heaters for low and moderate temperature applications. Solar Energy 1963;5(3):117–24.
[49] Duffie JA, Beckmann WA. Solar energy thermal processes. NY: Wiley; 1974.
[50] Treybal RE. Mass transfer operations, 3rd ed., NY: McGraw-Hill; 1980.
[51] Kreith F, Bohem RF. Direct-contact heat transfer. Washington: Hemisphere Pub. Corp.; 1988.
[52] Younis MA, Darwish MA, Juwayhel F. Experimental and theoretical study of a humidification–dehumidification desalting system. Desalination 1993; 94:11–24.
[53] Ben-Amara M, Houcine I, Guizani A, Maalej M. Experimental study of a multiple-effect humidification solar desalination technique. Desalination 2004;170:209–21.
[54] El-Agouz SA, Abugderah M. Experimental analysis of humidification process by air passing through seawater. Energy Conversion and Management 2008;49(12):3698–703.
[55] Lydersen AL. Mass transfer in engineering practice. NY: John Wiley & Sons; 1983.
[56] Orfi J, Laplante M, Marmouch H, Galanis N, Benhamou B, Nasrallah SB, et al. Experimental and theoretical study of a humidification–dehumidification water desalination system using solar energy. Desalination 2004;168:151.

[57] Wallis JS, Aull RJ. Improving cooling tower performance. Hydrocarbon Engineering 1999;(May):92–5.
[58] Mirsky GR, Bauthier J. Evolution of cooling tower fill. CTI Journal 1993;14(1):12–9.
[59] Aull RJ, Krell T. Design features of cross-fluted film fill and their effect on thermal performance. CTI Journal 2000;21(2):12–33.
[60] Kloppers JC. A critical evaluation and refinement of the performance prediction of wet-cooling towers. PhD dissertation. University of Stellenbosch; 2003.
[61] Kroger DG. Air-cooled heat exchangers and cooling towers thermal-flow performance evaluation and design, vols. I and II. Tulsa, Okla: Penwell Corp.; 2004.
[62] ASHRAE handbook: fundamentals. Society of Heating, American, Refrigerating, Air-Conditioning Engineers, and Inc., ASHRAE; 2005.
[63] Farid MM, Parekh S, Selman JR, Al-Hallaj S. Solar desalination with humidification–dehumidification cycle: mathematical modeling of the unit. Desalination 2002;151:153–64.
[64] Nawayseh NK, Farid MM, Al-Hallaj S, Tamimi AR. Solar desalination based on humidification process. Part I. Evaluating the heat and mass transfer coefficients. Energy Conversion Management 1999;40:1423–39.
[65] Bourouni K, Chaibi M, Martin R, Tadrist L. Applied Energy 1999;64:129.
[66] Klausner JF, Li Y, Darwish M, Mei R. Innovative diffusion driven desalination process. ASME Journal of Energy Resources Technology 2004;126: 219–25.
[67] Farid MM, Al-Hajaj AW. Solar desalination with humidification–dehumidification cycle. Desalination 1996;106:427–9.
[68] Li Y, Klausner JF, Mei R, Knight J. Direct contact condensation in packed beds. International Journal of Heat and Mass Transfer 2006;49:4751–61.
[69] Li Y, Klausner JF, Mei R. Performance characteristics of the diffusion driven desalination process. Desalination 2006;196:188–209.
[70] Threlkeld JL. Thermal environmental engineering, 2nd ed., Prentice-Hall Inc.; 1970. pp. 254–65.
[71] Pacheco-Vega A, Diaz G, Sen M, Yang KT, McClain RL. Heat rate prediction in humid air–water heat exchangers using correlations and neural networks. ASME Journal of Heat Transfer 2001;123:348–54.
[72] McQuiston FC. Heat, mass and momentum transfer data for five plate-fin tube heat transfer surfaces. ASHRAE Transactions 1978;84(Part 1):266–93.
[73] McQuiston FC. Correlation for heat, mass and momentum transport coefficients for plate-fin tube heat transfer surfaces with staggered tubes. ASHRAE Transactions 1978;84(Part 1):294–309.
[74] Beckmann JR. Method and apparatus for simultaneous heat and mass transfer utilizing a carrier gas. US Patent No. 6,911,121 (2005).
[75] Beckmann JR. Dew-vaporation desalination 5,000-gallon-per-day pilot plant. Desalination and water purification research and development program report no. 120; 2008.
[76] Klausner JF, Mei R. Diffusion driven desalination apparatus and process. US Patent No. 6,919,000 (2005).
[77] Khedr M. Techno-Economic Investigation of an Air humidification–dehumidification Desalination Process. Chemical Engineering Technology 1993;16: 270–4.
[78] Wahlgren RV. Atmospheric water vapor processor designs for potable water production: a review. Water Research 2001;35:1–22.
[79] Narayan GP, Elsharqawy MH, Lienhard JH, Zubair SM. Thermodynamic analysis of humidification dehumidification desalination cycles; Accepted for publication in Desalination and Water Treatment, 2009.
[80] Al-Hallaj S, Farid MM, Tamimi AR. Solar desalination with humidification–dehumidification cycle: performance of the unit. Desalination 1998;120:273–80.
[81] Garg HP, Adhikari RS, Kumar R. Experimental design and computer simulation of multi-effect humidification (MEH)–dehumidification solar distillation. Desalination 2002;153:81–6.
[82] Nafey AS, Fath HES, El-Helaby SO, Soliman AM. Solar desalination using humidification–dehumidification processes. Part II. An experimental investigation. Energy Conversion Management 2004;45(7–8):1263–77.
[83] Al-Enezi G, Ettouney HM, Fawzi N. Low temperature humidification–dehumidification desalination process. Energy Conversion and Management 2006;47:470–84.
[84] Dai YJ, Zhang HF. Experimental investigation of a solar desalination unit with humidification and dehumidification. Desalination 2000;130:169–75.
[85] Dai YJ, Wang RZ, Zhang HF. Parametric analysis to improve the performance of a solar desalination unit with humidification and dehumidification. Desalination 2002;142:107–18.
[86] Yamali C, Solmus I. A solar desalination system using humidification–dehumidification process: experimental study and comparison with the theoretical results. Desalination 2008;220:538–51.
[87] Sharqawy MH, Lienhard VJH, Narayan GP, Zubair SM. Water separation under reduced pressure. US Patent. 2009; # USSN 12/554,726.
[88] Narayan GP, Sharqawy MH, Lienhard VJH, Zubair, SM. Water separation under varied pressure. US Patent. 2009; # USSN 12/573,221.
[89] Mistry KH, Lienhard VJH, Zubair SM. Second Law Analysis of Humidification-Dehumidification Desalination Cycles; paper under review, 2009.
[90] Sharqawy MH, Lienhard VJH, Zubair SM. Thermophysical properties of seawater: A review of existing correlations and data, Accepted for publication in Desalination and Water Treatment, 2009.